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HEAT TRANSFER BY DIRECT FLAME CONTACT FIRE TESTS - PHASE I

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Prepared for

National Academy of Sciences
Committee on Hazardous Materials
U.S. Coast Guard Cargo Containment Panel
Washington, D. C. 20418

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The laboratory data showed a maximum radiant heat transfer rate of 31,000 Btu/hr-ft^2 for JP-4 fires, with a flame thickness of about 15 ft required to obtain the maximum rate. The convective heat transfer coefficient was about 5 Btu/hr-ft^2-°F for the laboratory-scale heat transfer probe, and the average "flame temperature" was about 2450°F. The convective coefficient can be extended to other situations by standard techniques.

Several tests were run to determine the effects of intumescent paint on the temperature rise of metal specimens exposed to radiant heating.

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INTRODUCTION

This report presents the results of a laboratory study on the heat transfer rate from fires by direct flame contact. The work was undertaken to provide preliminary design information for several proposed large scale fire tests in which liquid storage tanks of up to 10,000 gal capacity would be subjected to direct flame contact heating tests. The ultimate goal is to be able to predict the heating rate of the tank and its contents for the design of vents to relieve the internal tank pressure during fires and thereby prevent rupture of the tank with subsequent loss of flammable or toxic contents.

The report first describes the simplified mathematical analysis of heat transfer from a flame to an object in direct contact with the flame. Specific examples are given for several cases of possible interest. The results of laboratory data on heat transfer from flames are then presented. It is shown how these data may be extended to other fire conditions. Laboratory tests on metal samples and samples coated with intumescent paints are reported. Finally, a brief test plan for large scale fire tests is presented.

HEAT TRANSFER MODELING

Consider an object to be surrounded by fire from a burning hydrocarbon fuel. The object is heated by the flame by radiation and convection. Each case of interest must be considered individually in order to evaluate its response to heating by the fire, but the fundamental information necessary to evaluate the heat transfer behavior involves only consideration of flame radiation, convective heat transfer, and the system properties. Three cases of interest will be considered here: heat transfer to a tank containing liquid, heat transfer to the metal above the liquid surface, and heat transfer to a metal object insulated on one side.

Heat Transfer to a Liquid-filled Tank

The tank is assumed to be full or nearly full of liquid. Since the metal wall of the tank will usually have a relatively high thermal conductivity, its resistance to heat transfer can be neglected. Likewise, the heat capacity of the metal wall will be small compared to the heat capacity of the liquid inside the tank, and it may also be neglected. The heat transfer equation may then be written as

$$q_r + q_c = q_t + q_e \qquad (1)$$

where q_r = radiant flux absorbed by the tank

q_c = convective flux to tank

q_t = heat transfer rate through tank wall

q_e = radiant flux emitted by tank

The convective heat flux to the tank can be given by

$$q_c = h(T_f - T) \qquad (2)$$

where h = convective heat transfer coefficient

T_f = effective "flame temperature"

T = tank wall temperature

The heat transfer through the tank wall is quite rapid because the internal wall is wetted by liquid. If the liquid contents of the tank are assumed to be heated uniformly (or if a suitable average temperature is used), the heat transfer through the tank wall is just the heat absorbed by the liquid as its temperature rises, and

$$q_t = MC_p \frac{dT}{dt} \qquad (3)$$

In Equation 3

M = mass of liquid in the tank

C_p = heat capacity of liquid

t = time

and the tank wall temperature and the temperature of the liquid in the tank are assumed to be equal.

The radiant flux emitted by the tank, q_e, is small in comparison to the other terms and can be neglected unless the tank temperature rises beyond 500°F. Equation 1 can be then written as

$$MC_p \frac{dT}{dt} = q_r + h(T_f - T) \quad (4)$$

and the temperature of the tank contents at any time can be calculated if the initial temperature is known. Equation 4 applies only as long as the tank contents remain below the liquid boiling point and no significant evaporation takes place.

In practice, q_r, h, and T_f are taken as constants. Generally speaking, the radiative flux will be greater than the convective flux by at least a factor of two or three, and the boiling temperature will limit the maximum value of T. Once boiling of the bulk liquid begins, the boiloff rate can be found from

$$m\ \Delta H_v = q_r + h(T_f - T_b) \quad (5)$$

where m = mass evaporation rate due to boiling

ΔH_v = heat of vaporization of liquid

T_b = liquid boiling temperature

The evaporation rate can be used to calculate the pressure rise in the tank and subsequently to calculate the venting time and venting rate for the tank.

Heat Transfer to a Gas-filled Tank

If a tank is full of gas when exposed to a fire, the tank wall temperature will rise because the heat transfer rate to the gas inside the tank is relatively slow. The heat transfer equation may be written as

$$q_r + q_c = q_t + q_e + q_g \quad (6)$$

where q_t is now taken to be the heat transfer rate to the tank wall and q_g is the convective heat flux to the gas in the tank. The convective flux from the fire is

$$q_c = h(T_f - T) \tag{7}$$

where T is the tank wall temperature. The emitted flux is

$$q_e = \varepsilon\sigma T^4 \tag{8}$$

where ε = average emittance at temperature T

σ = Stephan-Boltzmann constant

(In reality, emission includes two terms, emission from the outer surface of the tank and emission from the inner surface of the tank. However, for a non-absorbing gas the inner surface receives the same net radiative flux from its surroundings as it emits, thus canceling the effect of the emitted flux on the interior. If the tank contains an absorbing gas or liquid, adjustments must be made in the heat transfer equation.)

The heat absorbed by the tank wall is

$$q_t = \rho_t C_t x \frac{dT}{dt} \tag{9}$$

where ρ_t = density of tank wall

C_t = heat capacity of tank wall

x = tank wall thickness

Heat transfer to the gas inside the tank is

$$q_g = h_g\ (T - T_g) \tag{10}$$

where h_g = convective heat transfer coefficient for inside tank wall

T_g = gas temperature inside tank

For a transparent gas, the temperature rise rate will be relatively slow, so T_g can be taken to be a constant.

Combining Equations 6 through 10 and rearranging the result gives

$$\rho_t C_t x \frac{dT}{dt} = q_r + h(T_f - T) - \varepsilon\sigma T^4 - h_g(T - T_g) \qquad (11)$$

Equation 11 can be used for calculating the tank wall temperature up to the time at which the tank wall begins to melt. The tank wall will obviously fail at the melting point if it has not failed previously.

Heat Transfer to an Uninsulated Object

The heat transfer rate to an uninsulated object inside a fire can be used to predict the temperature rise of such an object. Such calculations are useful in predicting the failure time of tank supports, for example. The basic equation for heat transfer is

$$M_s C_s \frac{dT}{dt} = q_r + h(T_f - T) - \varepsilon\sigma T^4 \qquad (12)$$

where M_s is the mass of the object and C_s is its heat capacity. The object is assumed to have a high thermal conductivity, such as for a metal.

The size and geometry of the object are important in determining the range of temperature rise. For example, consider three cases, a large flat plate, a long cylinder, and a sphere.

For a flat plate, heat transfer through the edges of the plate can be neglected. Then

$$x_s \rho_s C_s \frac{dT}{dt} = 2q_r + 2h(T_f - T) - 2\varepsilon\sigma T^4 \qquad (13)$$

where the flame is assumed to contact both sides of the plate. The plate thickness is x_s and its density is ρ_s. If exposure is long enough and the object is not melted or consumed by the fire, the steady state temperature can be calculated from

$$q_r + h(T_f - T) = \varepsilon\sigma T^4 \qquad (14)$$

If the object is a long cylinder, heat transfer to the cylinder through its ends may be neglected, and

$$\frac{d\rho_s C_s}{4} \frac{dT}{dt} = q_r + h(T_f - T) - \varepsilon\sigma T^4 \qquad (15)$$

where d is the diameter of the cylinder. Notice that in this case the rate of temperature rise is inversely proportional to the diameter of the object, assuming all other parameters to be constant.

For a sphere of diameter d the rate of temperature rise is

$$\frac{d\rho_s C_s}{6} \frac{dT}{dt} = q_r + h(T_s - T) - \varepsilon\sigma T^4 \qquad (16)$$

where d is the diameter of the sphere. The rate of temperature rise for the sphere is also inversely proportional to its diameter. The rate of rise of a sphere is 1.5 times as rapid as for a cylinder because the ratio of surface area to volume is 1.5 time as large.

The steady state temperature of both cylinders and spheres can be calculated from Equation 14, just as for a flat plate.

None of the above equations account for melting of the object during the heating process. If the metals have a constant melting point, the fraction of metal melted can be calculated by substituting

$$\rho_s C_s \frac{dT}{dt} = \rho_s \Delta H_f \frac{df}{dt} \qquad (17)$$

in Equation 13, 15, or 16 when the temperature reaches the melting point, providing the solid is contained as it melts. The temperature is taken as the melting temperature, f is the fraction melted, and ΔH_f is the heat of fusion. If melting occurs over a temperature range, as found with some alloys, the temperature rise must continue to be calculated during melting. The substitution

$$\rho_s C_s \frac{dT}{dt} = \rho_s C_s + \frac{\rho_s \Delta H_f}{(T_E - T_B)} \qquad (18)$$

may be used to estimate the temperature. Equation 18 is based on a linear estimate of the fraction of melting as a function of temperature. The fraction melted may be calculated from

$$f = \frac{T - T_B}{T_E - T_B} \qquad (19)$$

where T_B is the temperature at which melting begins, and T_E is the temperature at which melting is completed.

LABORATORY HEAT TRANSFER DATA

Most of the independent parameters appearing in the heat transfer equations in the preceding section are properties of the materials subjected to the fire. Three of the parameters are properties of the fire, and the values of the parameters may depend on the fuel being burned, the fire size, and the geometry of the object receiving heat from the fire. These parameters are the radiative flux , the convective heat transfer coefficient in the fire, and the flame "temperature." A series of laboratory measurements was made to determine the values of these parameters for heat transfer calculations.

Two types of laboratory tests were run, one in which radiant heat transfer from a fire was measured for various fire thicknesses, and one in which the total heat transfer rate was measured for a probe completely surrounded by fire.

Radiative Heat Transfer Experiments

Figure 1 is a schematic diagram of the experimental apparatus used for measuring radiative heat transfer properties. An open liquid-filled burner about 20 inches long and 2 inches wide was filled with JP-4. The fuel was maintained at a constant level in the burner by a liquid-sealed constant head siphon which led to a fuel supply tank. The burner and radiometer were contained inside a cabinet which was open beneath and above the burner. Fresh air was drawn up through the cabinet, passing through a flow straightening

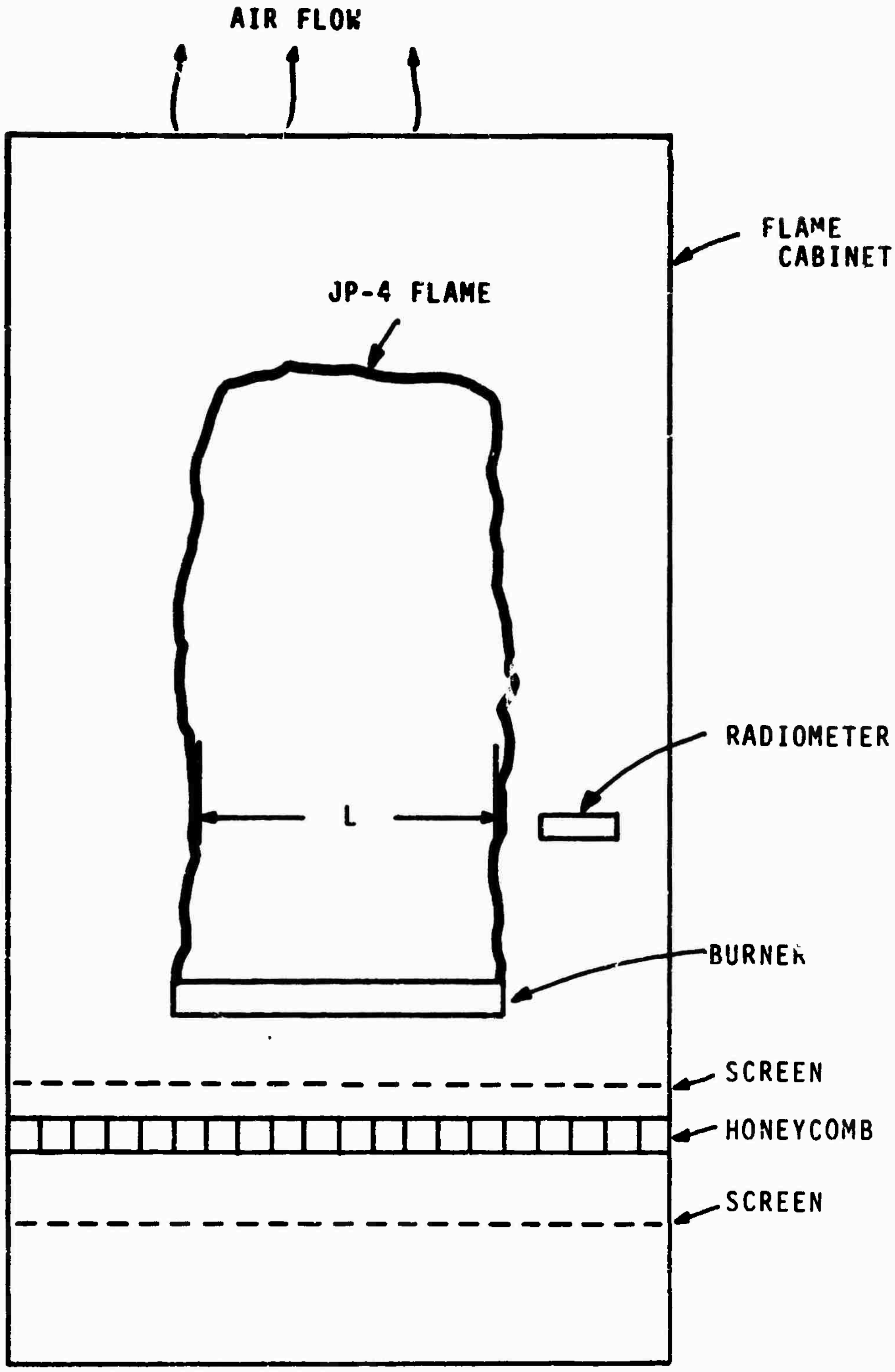

Figure 1. Schematic Diagram of Laboratory Radiation Measuring Apparatus.

section, then past the flame, and finally out the top of the cabinet. Combustion products were exhausted through the top of the cabinet along with the air being drawn through the cabinet. The flame burned uncontrolled above the burner except for restrictions on the radiometer path length (L in Figure 1), which were made in order to determine the radiation properties. The scale of turbulence was smaller than that of the direct flame contact experiments, which, as will be shown later, influences the opacity of the flame. There was no premixing of air and fuel; all air entered the flame zone by turbulent mixing and ultimately by diffusion. As is characteristic with diffusion flames, the flame path length (L) decreased slightly for a few inches above the burner, then increased farther up the flame.

Radiometer measurements were made high enough up into the flame that the fuel and air were well mixed. Just above the burner surface, the flame had a fuel-rich zone which was visually transparent. This zone was avoided during radiometer measurements. The radiometer was designed for low net fluxes and was fitted with a water-cooled view restrictor to limit its viewing angle. Both radiometer and view restrictor could be purged with an inert gas during operation to prevent accumulation of smoke or soot. The radiometer sensor was protected by a sapphire window to prevent convective effects. The sapphire window transmits radiation from the visible region to more than 6 microns in wavelength, so that essentially all of the flame radiation energy reaches the sensor. The radiometer was calibrated with the view restrictor in place using a blackbody radiation source.

Traditionally, radiant heat transfer from a flame has been calculated using a modification of the Stephan-Boltzmann law,

$$q_r = F\varepsilon_f \sigma \, T_f^{\,4} \, (1 - e^{-\kappa L}) \qquad (20)$$

where q_r = radiant flux

ε_f = emittance of the flame

σ = Stephan-Boltzmann constant

T_f = flame "temperature"

κ = flame opacity coefficient

L = flame thickness

F = view factor

In practice, Equation 20 is applied by first measuring the radiant flux from a fire and then calculating from that flux an "equivalent blackbody temperature." The emittance is then assumed to be unity and the equivalent blackbody temperature is used for heat transfer predictions. Alternatively, the flame "temperature" is measured by an independent technique, such as an optical pyrometer or a thermocouple, and the emittance is calculated from the measured radiant flux and the temperature.

It is well known that the Stephan-Boltzmann law does not apply to a flame because the flame emits radiation both in a continuum and in bands. The continuum radiation is attributed to hot carbon particles and the major bands are attributed to emission from hot H_2O and CO_2 molecules. Love (8) has proposed that the flame be treated as an emitting, absorbing medium, with the basic transport equation

$$\frac{dI_\lambda}{dL} = J_\lambda - \beta_\lambda I_\lambda \tag{21}$$

where I = monochromatic intensity

J_λ = monochromatic volumetric emission coefficient

β_λ = monochromatic extinction coefficient

L = path length

Equation 21 is one-dimensional, and has the solution

$$I_\lambda = \frac{J_\lambda}{\beta_\lambda}(1 - e^{-\beta_\lambda L}) \tag{22}$$

Further integration over the geometry of the particular problem of interest and the wavelengths containing significant flame radiant energy can be made if enough information on the values of J_λ and β_λ can be obtained.

Unfortunately, measured values of J_λ and β_λ are not available for the fuels of interest in this study. However, Equation 22 can be integrated over all applicable wavelengths and geometry to obtain a simplified equation which can be used for predicting heat transfer. The simplified form is

$$q_r = FA(1 - e^{-bL}) \tag{23}$$

where A is the maximum emission from the fire and b is an extinction coefficient. A comparison of Equations 20 and 23 shows that they are equal if

$$b = \kappa \tag{24}$$

and

$$A = \varepsilon_f \sigma T_f^4 \tag{25}$$

Both A and b are related to Equation 22; A is an "average" value of J_λ/β_λ, integrated over all important wavelengths, and b is an "average" value of β_λ, again integrated over all wavelengths. The view factor, F, is related to the geometry and converts intensity to total flux.

The radiation measurements made with the narrow angle radiometer were analyzed using Equation 23. The viewing angle of the radiometer was narrow enough so that only the flame was viewed. Measurements of the radiant flux from the fire were made for several path lengths through JP-4 flames by decreasing the flame length. Figure 2 shows the results. The data shown in Figure 2 were used to calculate the maximum flux, A, and the extinction coefficient, b, for the test fires. The results showed

$$A = 31{,}000 \text{ Btu/hr-ft}^2$$

$$b = 0.156 \text{ in}^{-1}$$

The value of b measured during the narrow angle radiometer tests is high. An optically thick fire, i.e., a fire which would emit 99 percent of the maximum radiant flux, would be only about 30 inches thick, a value significantly lower than that accepted by most investigators. The reason for the abnormally high value of b and the small optically thick fire is the small scale of turbulence in the test fires. With a burner only 2 inches wide, the size of individual flame zones is also small, but the number of flame interfaces is relatively large. Since emission and extinction occur primarily at the flame interfaces (where fuel and air are in the proper proportions for combustion), the extinction coefficient becomes larger as the scale of turbulence becomes smaller. Another way of explaining the large value of b due to

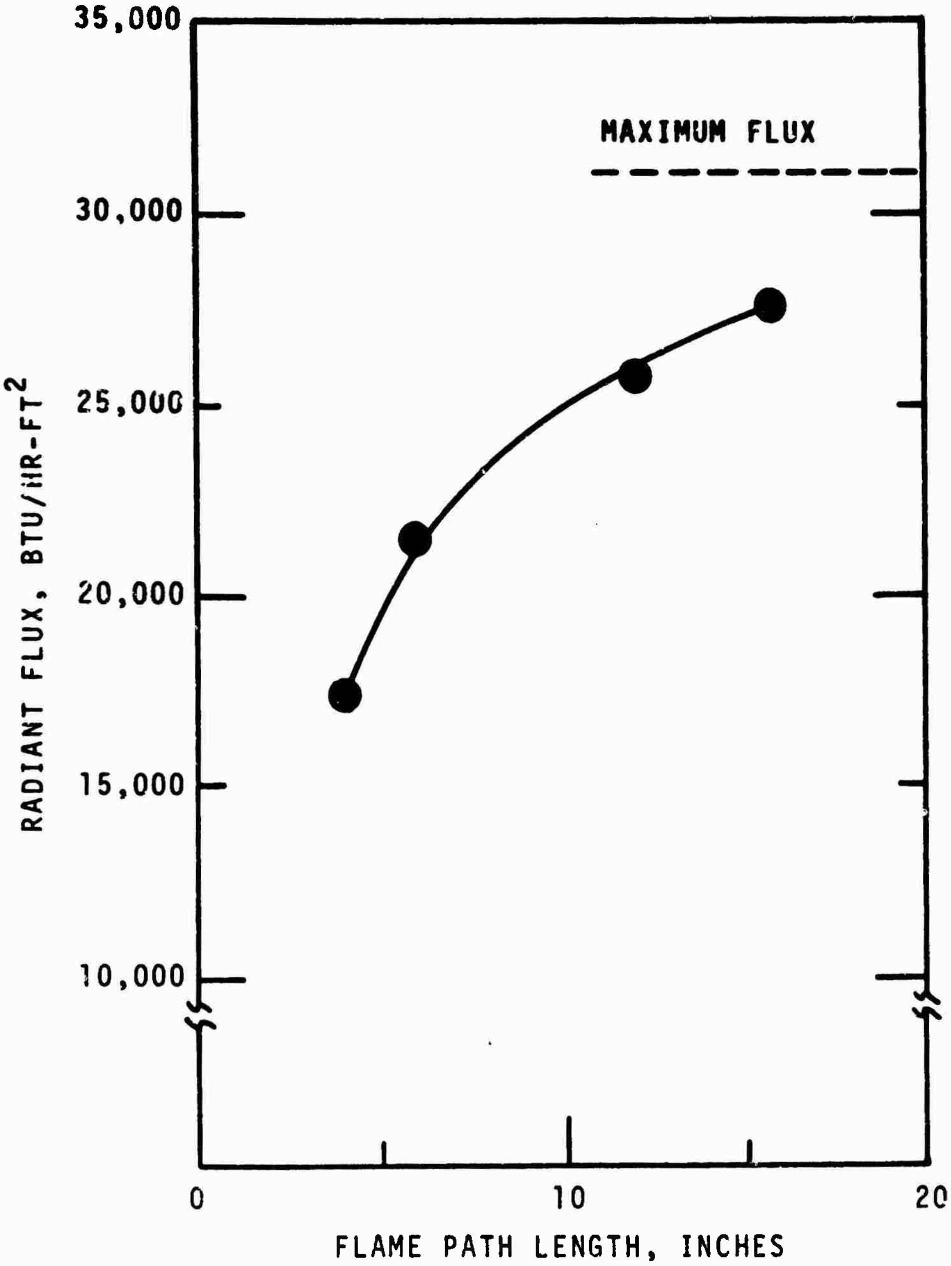

Figure 2. Radiant Flux for JP-4 Fires from a 2-Inch Wide Burner.

the smaller scale of turbulence is that the average concentration of flame zones is higher for a smaller scale of turbulence.

Since the value for b measured for the 2-inch burner was clearly not suitable for use in predicting heat transfer from moderate fires with roughly circular or square bases, radiation measurements were made on circular fires 12, 18, and 24 inches in diameter. Figure 3 shows a schematic diagram of the circular fire testing equipment, which was used in conjunction with the direct flame contact heat transfer experiments. A wide angle radiometer was mounted outside the flame zone but near the flame where the radiant flux was highest. The radiometer was water cooled and purged with nitrogen to prevent smoke accumulation on the sapphire window that covered the sensor. The view angle of the radiometer was greater than 150 degrees, so the entire field of view was not always covered by the flame. Figure 4 is a photograph of an 18-inch JP-4 fire.

Equation 23 was rearranged to

$$b = (-1/L) \ln (1 - q_r/FA) \tag{26}$$

in order to solve for the extinction coefficient for the circular burner fires. The value of 31,000 Btu/hr-ft^2, as determined from the narrow angle radiometer tests, was used for A, and view factors were taken from the work of Rein et al. (11). The flame diameters at the radiometer height varied from about 17 inches to about 32 inches, as measured from photographs of the flames.

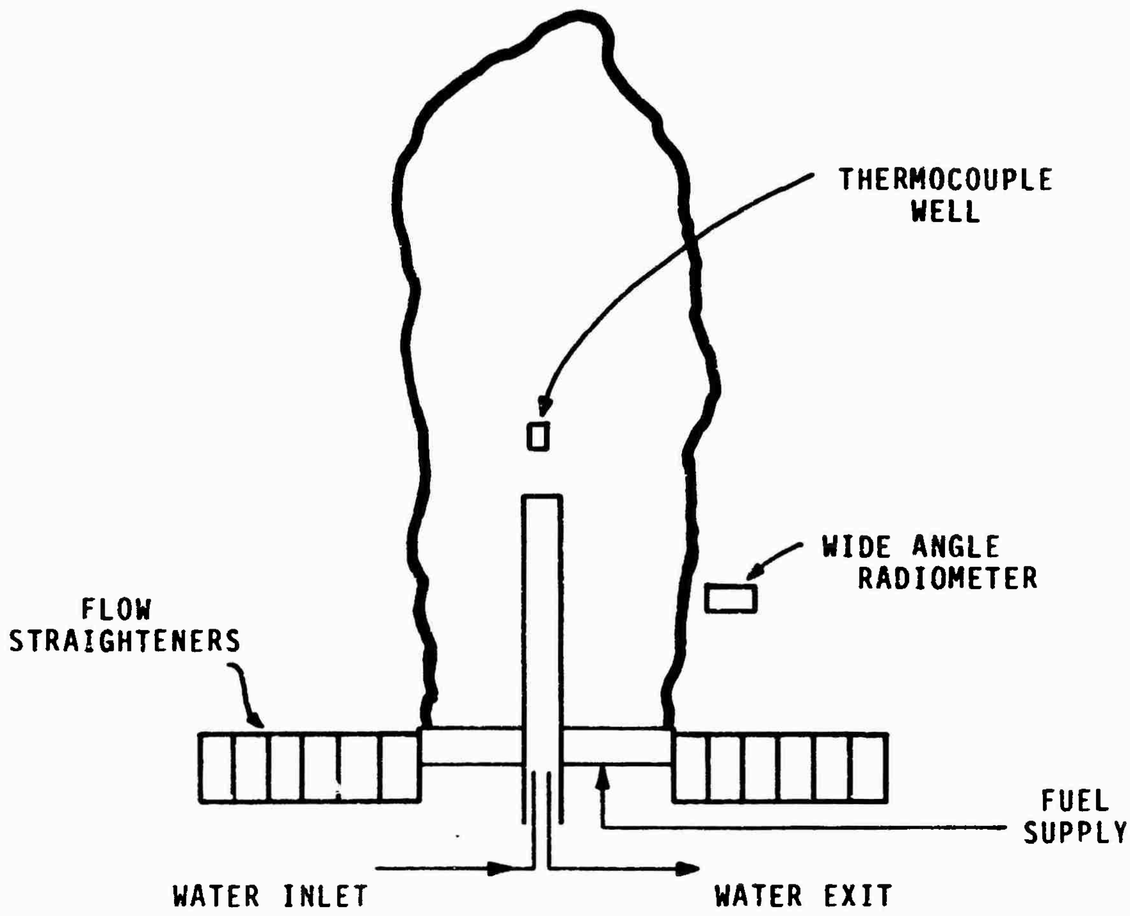

Figure 3. Schematic Diagram of Direct Flame Contact Heat Transfer Apparatus.

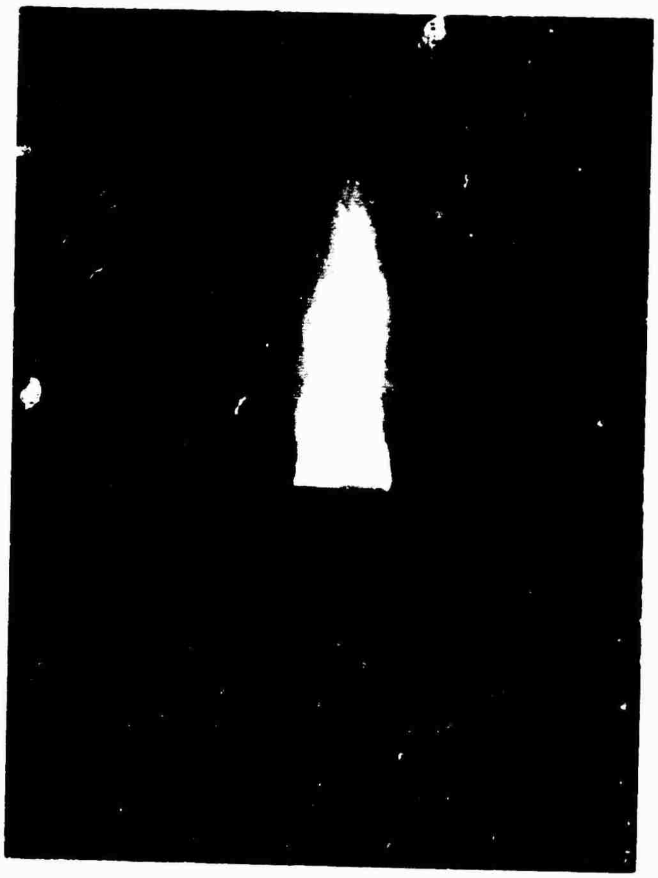

Figure 4. Photograph of an 18-Inch JP-4 Fire.

Figure 5 is a plot of the radiation data from the circular burner fires, plotted on semilogarithmic coordinates. Using the maximum flux of 31,000 Btu/hr-ft^2 found from the tests described above, the average extinction coefficient was found to be 0.0186 inch^{-1}. A fire which would emit 99 percent of the maximum flux based on this value of b would have to be about 20 ft through the flame zone. This value of b is within the range normally expected for hydrocarbon fires.

Convective Heat Transfer Experiments

Nine experiments were run using a water-cooled heat transfer probe. The probe was mounted at the center of a circular pan as shown schematically in Figure 3. Pan diameters were 12, 18, and 24 inches. Figure 6 shows the heat transfer probe, the thermocouple well for measuring the flame temperature, and the radiometers for measuring the radiant flux outside the fire. The flow straightening tubes surround the entire burner and test equipment. Figure 7 gives the dimensions of the probe and shows the location of the thermocouples on the outer wall of the probe.

The probe was constructed of three concentric tubes, as shown in Figure 8. Cooling water flowed up through the inner tube and back down and out between the inner tube and the intermediate tube. The space between the intermediate tube and the outer tube was filled with insulation. The purpose of the insulation was to decrease the heat transfer rate to the water and raise the temperature of the outer tube, which was in direct contact with the surrounding fire. It was hoped that the higher temperature of the

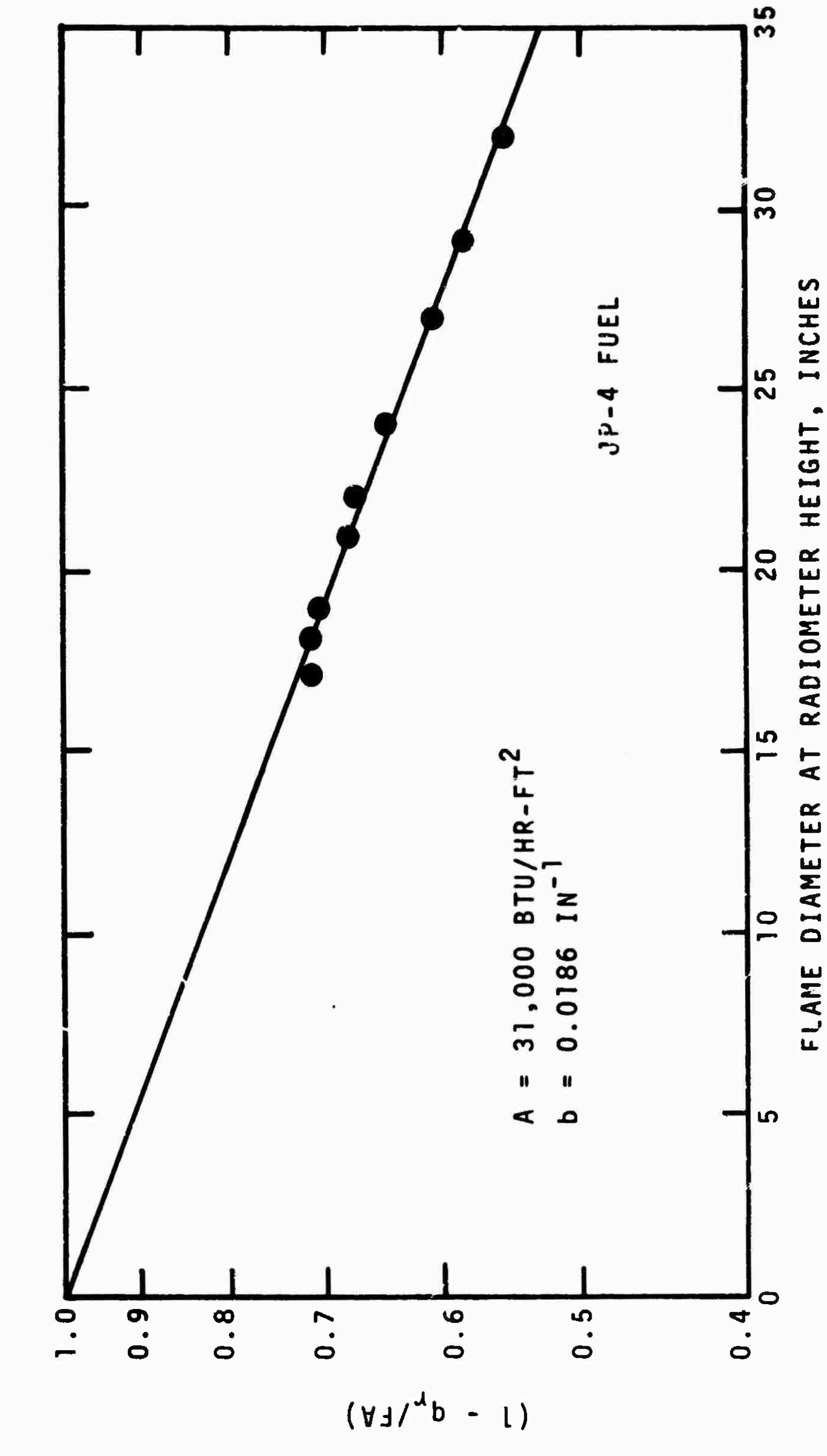

Figure 5. External Radiant Heat Transfer Data for Circular JP-4 Fires.

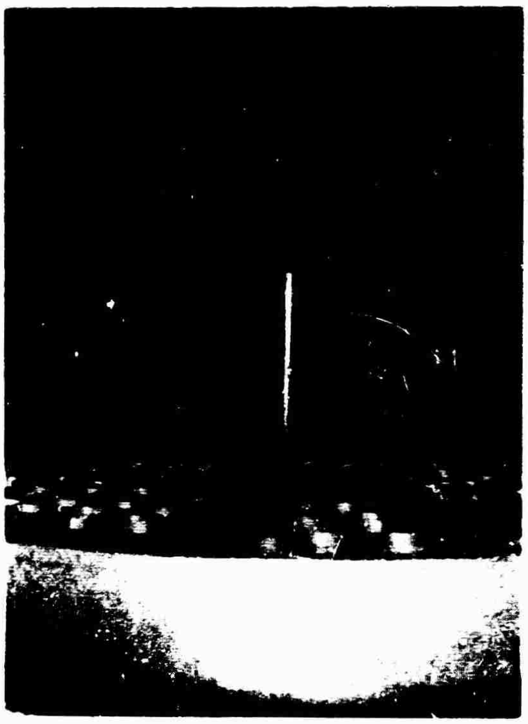

Figure 6. Heat Transfer Probe, Thermocouple Well, and Radiometers.

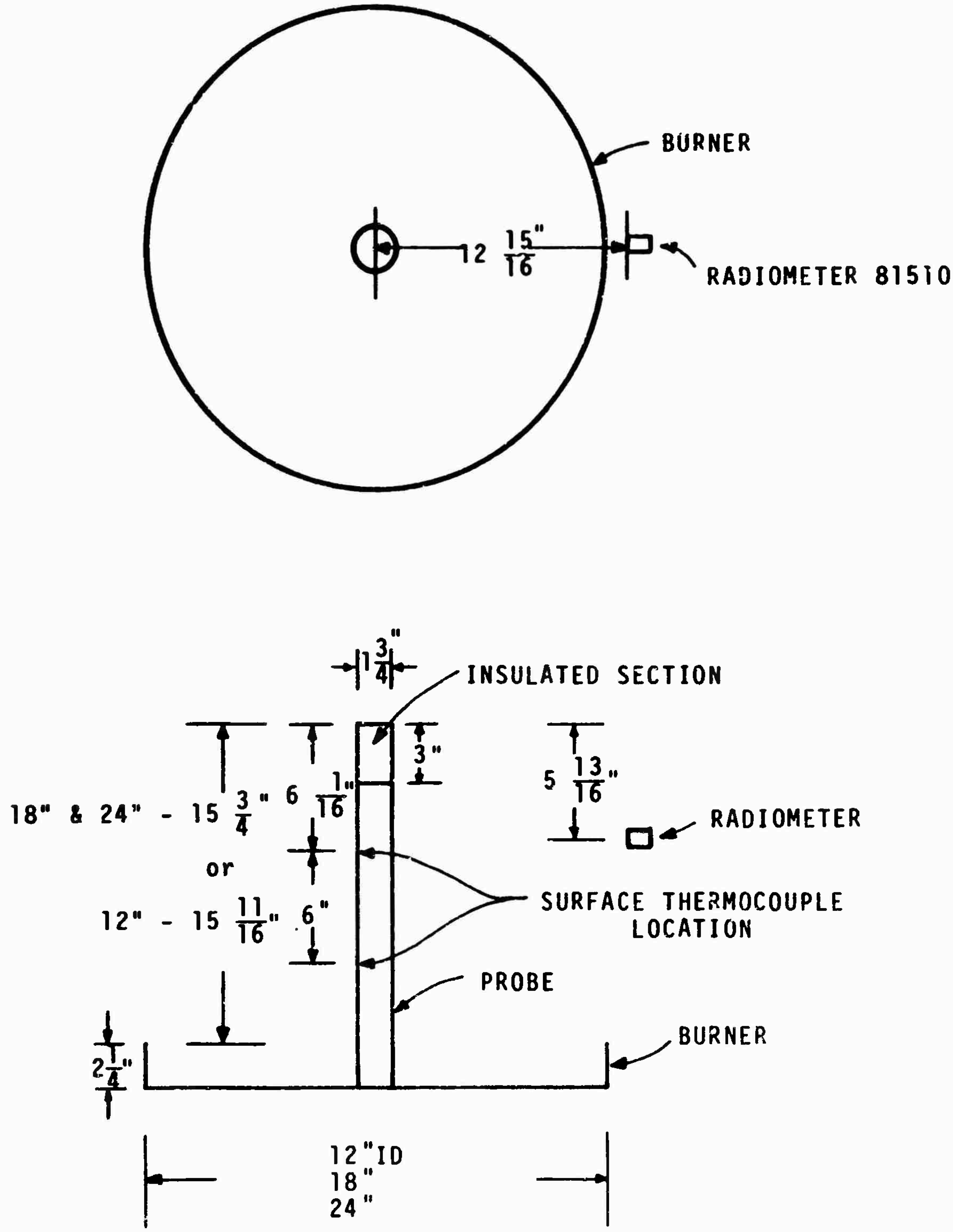

Figure 7. Details of Heat Transfer Probe.

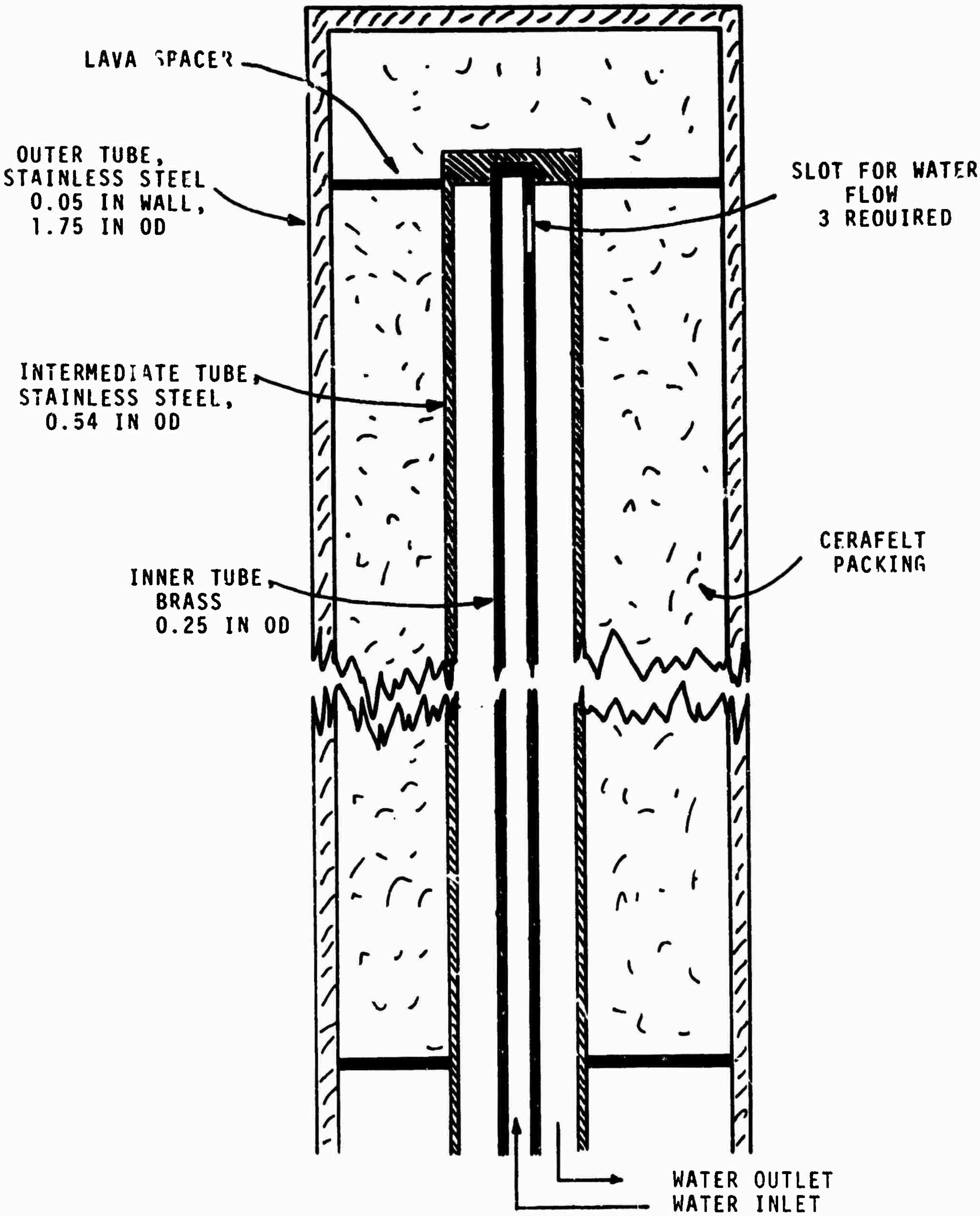

Figure 8. Schematic Diagram of Heat Transfer Probe.

outer wall would aid in preventing carbon deposition on the probe surface. However, even at the high temperatures at which the outer shell operated, more than 1100°F, some carbon still collected on the probe surface. The water flow rate was held constant during a test, and the temperatures of the inlet water, the outlet water, the probe wall, and the thermocouple well were monitored continuously.

A heat balance on the active area of the probe was written to determine the heat transfer rates. At steady state conditions,

$$q_r + q_c = q_w + q_e \tag{27}$$

where q is the heat flux absorbed by the water flowing through the probe, i.e.,

$$q_w = \frac{mC_w\Delta T_w}{A_p} \tag{28}$$

In Equation 28 m = mass flow rate of water

C_w = specific heat of water

T_w = temperature rise of the water flowing through the probe

A_p = active surface area of the probe, based on flame contact area

Since all the independent variables in Equation 28 are known, q_w can be calculated directly.

The emitted energy, q_e, is calculated from the Stephan-Boltzmann law,

$$q_e = \sigma T_p^4 \tag{29}$$

where T_p is the probe surface temperature. The emittance of the probe surface has been assumed to be unity because the surface is covered with a thin layer of carbon as soon as it is contacted by the fire.

The radiant flux, q_r, is calculated from Equation 23 with the view factor taken as unity because the entire probe is surrounded by flame. The optical path length is obtained by taking the average distance from the probe wall to the outer edge of the flame. The flame dimensions are obtained from photographs taken during the experiments. The values of A and b for the radiant heat transfer calculations are those measured as described in the previous section.

Once q_w, q_e, and q_r have been obtained, q_c can be found from Equation 27. The convective heat transfer rate involves both the flame temperature and a convective heat transfer coefficient, i.e.,

$$q_c = h(T_f - T_p) \tag{30}$$

Since it is desired to determine the convective coefficient, the flame temperature must be determined. The term "flame temperature" is difficult to define because a flame is a reacting medium and is not in equilibrium. Several techniques have been used in the past to determine a "flame temperature," but the results vary widely, especially for turbulent diffusion flames. For the purposes of

heat transfer, the flame temperature is best determined from heat transfer data. In the present experiments, the flame temperature was determined by a steady state heat balance on a small stainless steel cylinder suspended in the flame near the heat transfer probe. At steady state, the heat transfer equation for the cylinder is

$$q_r + h_c (T_f - T_c) = \varepsilon\sigma T_c^4 \qquad (31)$$

where h_c is the convective heat transfer coefficient and T_c is the cylinder temperature. Knudsen and Katz (6) give correlations for estimating convective heat transfer coefficients, from which h_c was estimated to be about 10 Btu/hr-ft^2-°F. The steady state cylinder temperatures were measured during each of the experimental runs, and the emittance was taken to be one. The radiant flux was calculated in the same manner as for the larger heat transfer probe. All the parameters in Equation 31 were then known except for the flame temperature, which could be calculated.

Once the flame temperature had been calculated, Equation 27 could be used to calculate the average convective heat transfer coefficient for the probe. Table 1 shows the results and summarizes the data from the direct flame contact heat transfer measurements. The convective heat transfer coefficient obtained from the heat transfer data ranged from a minimum of 3.9 Btu/hr-ft^2-°F to a maximum of 9.4 Btu/hr-ft^2-°F. The value of 9.4 appears a little high. The average value of the convective coefficients, excluding the single data point which appears questionable, is approximately 5 Btu/hr-ft^2-°F. The average "flame temperature" is calculated to be about 2450°F.

TABLE 1

SUMMARY OF HEAT TRANSFER DATA

Nominal Burner Dia	Flame Dia	External Radiant Flux*	Radiant Flux	Flux Absorbed by Water	Cooled Probe Temp	Uncooled Cylinder Temp	Flame Temp (Calculated)	Convective Flux	Probe Convective Coefficient
			Heat Transfer Data for Interior of Flame						
inches	inches	$\frac{Btu}{hr\text{-}ft^2}$	$\frac{Btu}{hr\text{-}ft^2}$	$\frac{Btu}{hr\text{-}ft^2}$	°F	°F	°F	$\frac{Btu}{hr\text{-}ft^2}$	$\frac{Btu}{hr\text{-}ft^2}$
12	17	5640	6260	850	1150	1310	2360	6220	5.1
12	18	5640	6540	780	1150	1300	2310	5770	5.0
12	19	5850	6820	870	1150	1300	2270	5630	5.0
18	22	8490	7750	890	1180	1340	2340	4520	3.9
18	24	9110	8000	1600	1170	1360	2440	5860	4.6
18	21	8350	7310	930	1170	1340	2400	5820	4.7
24	27	12100	9020	1470	1360	1400	2570	11300	9.4
24	32	13700	9580	1100	1300	1480	2920	8140	5.0
24	29	12800	8550	1320	1320	1430	2790	10100	7.0

*Incident flux as measured by radiometer. Not corrected for view factor.

Discussion of Heat Transfer Data

The results of the direct flame contact heat transfer experiments on JP-4 fires are summarized as follows:

Maximum Radiant Heat Flux, A,	31,000 Btu/hr-ft^2
Flame Extinction Coefficient, b,	0.0186 in^{-1}
Convective Heat Transfer Coefficient, h,	5 Btu/hr-ft^2-°F
Average "Flame Temperature,"	2450 °F

The convective flux to a cool object in a fire will be about 10,000 Btu/hr-ft^2. For a large fire the convective flux is therefore about 25 percent of the total. For smaller fires (a foot or two in diameter), the convective flux may be greater than the radiative flux.

It is worthwhile to compare the present results with the results of previous investigators; their results are summarized in Table 2. The data of Bader (2) and Copley (3) are for maximum heat transfer rates, including both convection and radiation, although the authors consider convection to be negligible.

TABLE 2

MAXIMUM JP-4 HEAT FLUXES

Investigator	Maximum Flux Btu/hr-ft^2
Atallah and Allan (1)*	31,000
Bader (2)	36,000 - 48,000
Copley (4)	31,000
Law (7)	54,000
Neill, et al. (9)	31,000

*Based on blackbody radiation from a 1600°F flame.

The work of Atallah and Allan (1) is based on assumed blackbody radiation at 1600°F. (The justification for using 1600°F was not given.) Actually, the only way an "equivalent blackbody temperature" can be determined is by first measuring the radiant flux, and then calculating the temperature of a blackbody which would have the same flux. In view of the fact that it is generally easier just to use the flux, nothing is gained by calculating a temperature in this manner.

Bader's (2) heat fluxes were measured assuming that all the heat transfer from the JP-4 fire to the sensor was due to radiation. The measured fluxes were stated to be from about 36,000 Btu/hr-ft^2 to 48,000 Btu/hr-ft^2, although if calculations are made based on the data plots shown in the paper, the maximum flux would be about 36,000 Btu/hr-ft^2. Bader measured the temperature of small steel plates in the same fires, and these temperatures can be used in Equation 31 to calculate the flame temperature. If 36,000 Btu/hr-ft^2 is used as the radiant flux with the plate temperature of 1800°F given by Bader, the flame temperature will be calculated to be about 2650°F. The convective heat transfer to the radiation sensor can then be estimated to be about 7000 Btu/hr-ft^2, so that convective heating must be considered. Taking the maximum radiant flux from the present work, 31,000 Btu/hr-ft^2, and adding the convective flux of about 7,000 Btu/hr-ft^2 gives about 38,000 Btu/hr-ft^2, which compares very well with Bader's total fluxes on JP-4 fires 18 ft square (324 ft^2 area).

Copley (4) also considered the entire flux to be due to radiation. HIs fires were 10 ft by 18 ft, and if the flux to his heat transfer cylinder is calculated using Equations 23 and 3C and the heat transfer properties of the present work, the radiant flux is predicted to be about 28,000 Btu/hr-ft^2 and the convective flux is predicted to be about 4000 Btu/hr-ft^2, for a total flux of 32,000 Btu/hr-ft^2. This value is in excellent agreement with Copley's work.

Law (7) proposed that flame radiant fluxes be calculated assuming a blackbody flame temperature of 2012°F (1100°C). She then quotes the calculated radiant flux to be 54,000 Btu/hr-ft^2. Actually, a blackbody at 2012°F radiates at about 64,000 Btu/hr-ft^2, so there appears to be an error in Law's calculation. She quotes Rasbash et al. (10) and Duggan et al. (5) as measuring heat fluxes of 20,000 Btu/hr-ft^2 to 28,000 Btu/hr-ft^2, and says these values were for thin flames and that the experimental values are therefore consistent with her proposed theoretical value of 54,000 Btu/hr-ft^2. Law says that the emittance of the flames can be taken as unity if the flames are more than about 5 ft thick.

Neill et al. (9) predicted maximum heat transfer rates of about 31,000 Btu/hr-ft^2 for JP-4 fires based on data obtained using a cooled heat transfer probe in fires up to 18 in in diameter. They attributed about 7000 Btu/hr-ft^2 of the heat flux to convective heating and about 24,000 Btu/hr-ft^2 to radiative heating. However, in this study some difficulty was experienced with flame stability such that the flame did not always surround the heat transfer probe completely.

Atallah and Allan (1), quoting the results of Rasbash, et al. (10) and Burgess et al. (3), give values of the extinction coefficient for liquid hydrocarbon fires from about 0.03 inches^{-1} to 0.076 inches^{-1}. Neill et al. (9) measured the extinction coefficient to be 0.06 inches^{-1} for JP-4. These values differ from that obtained in the present study, 0.0186 inches^{-1}. The extinction coefficient is dependent on the flame turbulence, and the data of Neill et al. were taken for channel burner fires, which are long and narrow. The scale of turbulence is such that the extinction coefficient is larger than for fires where the scale of turbulence is larger. Since the maximum heat transfer rate by radiation is reached for a flame thickness of about 15 ft or less for any of the extinction coefficients, the difference is important primarily for small fires. In large scale fire tests, the flame thickness should be maintained at 15 ft or greater for maximum heat transfer rates.

Convective heat transfer has usually been neglected in measuring or predicting heat transfer rates inside fires. Most investigators have either ignored convection or lumped it together with radiation. The problem is that radiation and convection follow different scaling laws. They must therefore be calculated separately and then combined to give the total heat transfer rate. The convective heat transfer coefficient found in the present tests, 5 Btu/hr-ft^2-°F, is the same as that recommended by Neill et al. (9), which is the only other published value for heat transfer in the interior of a fire.

Knudsen and Katz (6) summarize several correlations for heat transfer to various solid bodies by convection. Two of those techniques for convective heat transfer to cylinders were used to predict heat transfer coefficients for comparison with that measured during the fire tests. One correlation is

$$\frac{dh}{k}_{T_{0.5}} = 0.615 \left(\frac{dU\rho}{\mu}\right)^{0.466}_{T_{0.5}} \qquad (32)$$

where d = cylinder diameter

k = thermal conductivity of gas

μ = gas viscosity

ρ = gas density

U = gas velocity

The Reynolds number must be between 40 and 4000, and all fluid properties are evaluated at temperature $T_{0.5}$, where

$$T_{0.5} = \frac{(T_w + T_\infty)}{2} \qquad (33)$$

In Equation 33, T_w is the cylinder wall temperature which corresponds to T_p in the fire tests, and T_∞ is the ambient temperature, which corresponds to T_f. Equation 32 was used to estimate the heat transfer coefficient for the probe under the conditions of the fire tests. The fluid properties were assumed to be those of air at the appropriate conditions of temperature and atmospheric pressure. The calculated convective heat transfer coefficient was 4 Btu/hr-ft^2-°F, which is about 20 percent lower than that measured during the fire tests. Equation 32 is based on data at turbulence intensities of

1 to 2 percent, and Knudsen and Katz (10) show data that indicate the heat transfer coefficient would be about 20 to 30 percent higher at turbulence intensities of 7 to 18 percent. The convective coefficient calculated from Equation 32 therefore compares very well with the measured value.

The second correlation Knudsen and Katz recommend is for heat transfer to cylinders at very high temperature differences. It is

$$\frac{hd}{k} = 0.6 \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{dU\rho}{\mu}\right)^{1/2} \left(\frac{T_w}{T}\right)^{0.12} \qquad (34)$$

where C_p is the heat capacity of the gas. In Equation 34 the fluid properties are evaluated at the bulk temperature and the Reynolds number must be between 300 and 2250. The heat transfer coefficient calculated for the conditions of the fire tests from Equation 34 is 4.9 Btu/hr-ft^2-°F, which agrees very well with the average value measured from the fire tests.

Based on the results of the fire tests and the data from the literature, the maximum radiant flux from a JP-4 fire can be taken as 31,000 Btu/hr-ft^2. The extinction coefficient depends on the scale of turbulence, but should not be much less than the value measured for the laboratory fires, 0.0186 inches^{-1}. In order to assure that the maximum radiant flux is produced during fire tests, the flame should be 15 to 20 ft thick. The convective heat transfer coefficient of 5 Btu/hr-ft^2-°F is applicable to the small probe used in the tests. For other probe sizes the correlations given

by Knudsen and Katz (6) should be used to estimate the convective coefficient. The "flame temperature" for these convective heat transfer calculations should be taken to be about 2450°F.

Other Laboratory Data

In addition to the data on basic flame heat transfer rates, several laboratory tests were run to check heat transfer models and to determine the usefulness of techniques for reducing heat transfer to objects subjected to flame heating. These tests consisted primarily of the measurement of the temperature rise of aluminum plates which were exposed to controlled radiant heat fluxes. The results for two of the tests in which the metal plates were painted with a single coat of flat black paint are shown in Figures 9 and 10. The data of Figure 9 are for an aluminum plate 0.25 inches thick exposed to a radiant flux of 9020 Btu/hr-ft^2. The calculated temperature line was obtained from the heat balance on the metal plate,

$$\rho_c x + \frac{\rho x \Delta H_f}{(T_E - T_B)} \frac{dT}{dt} = q_r - h(T - T_o) - \epsilon\sigma T^4 - \frac{k}{z}(T - T_o) \tag{35}$$

where ΔH_f = heat of fusion of the metal

T_B = temperature at beginning of melting

T_E = temperature at end of melting

k = thermal conductivity of insulation

z = thickness of insulation

T_o = ambient temperature

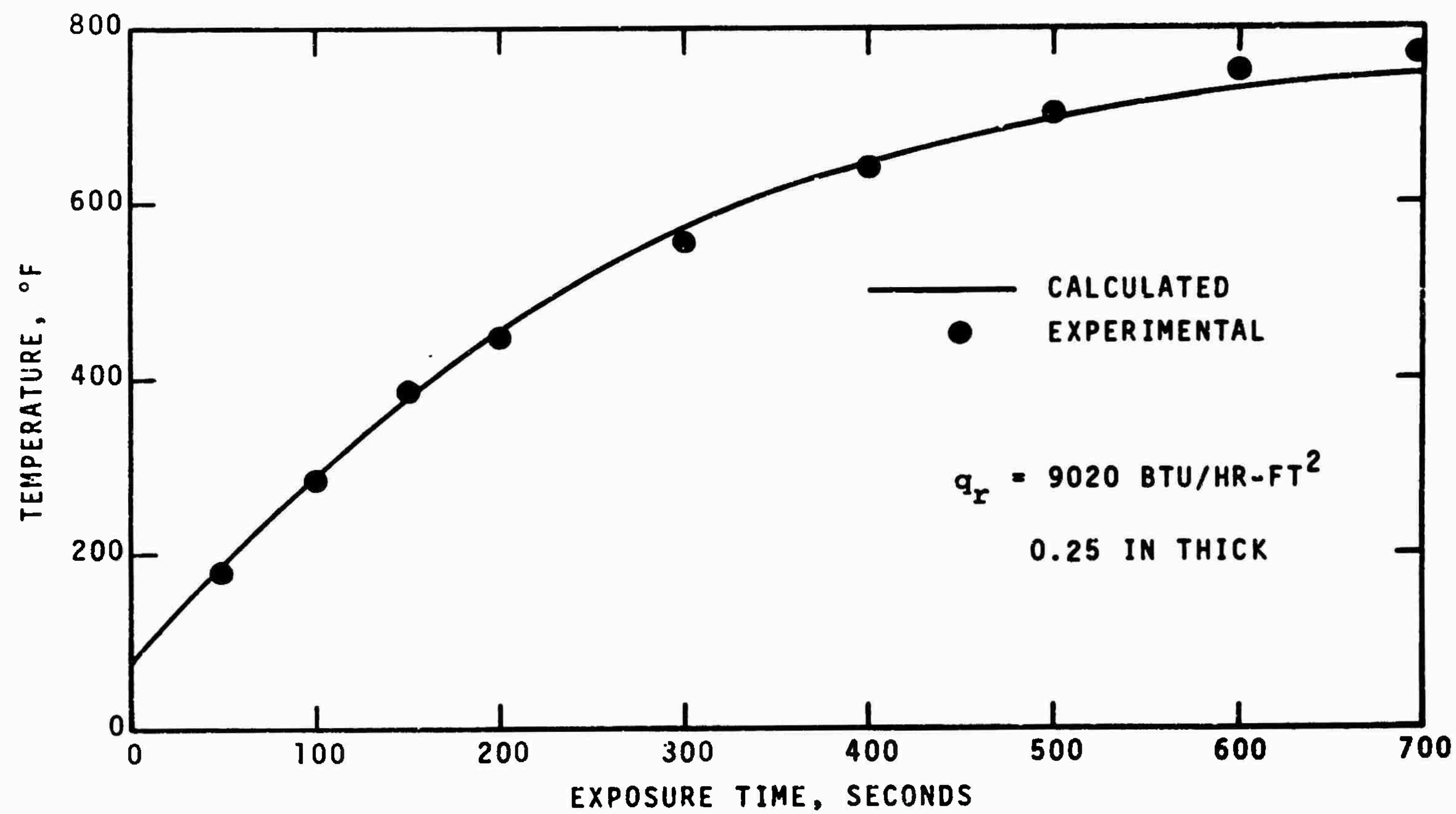

Figure 9. Calculated and Experimental Temperature Rise for 0.25-Inch Thick Aluminum.

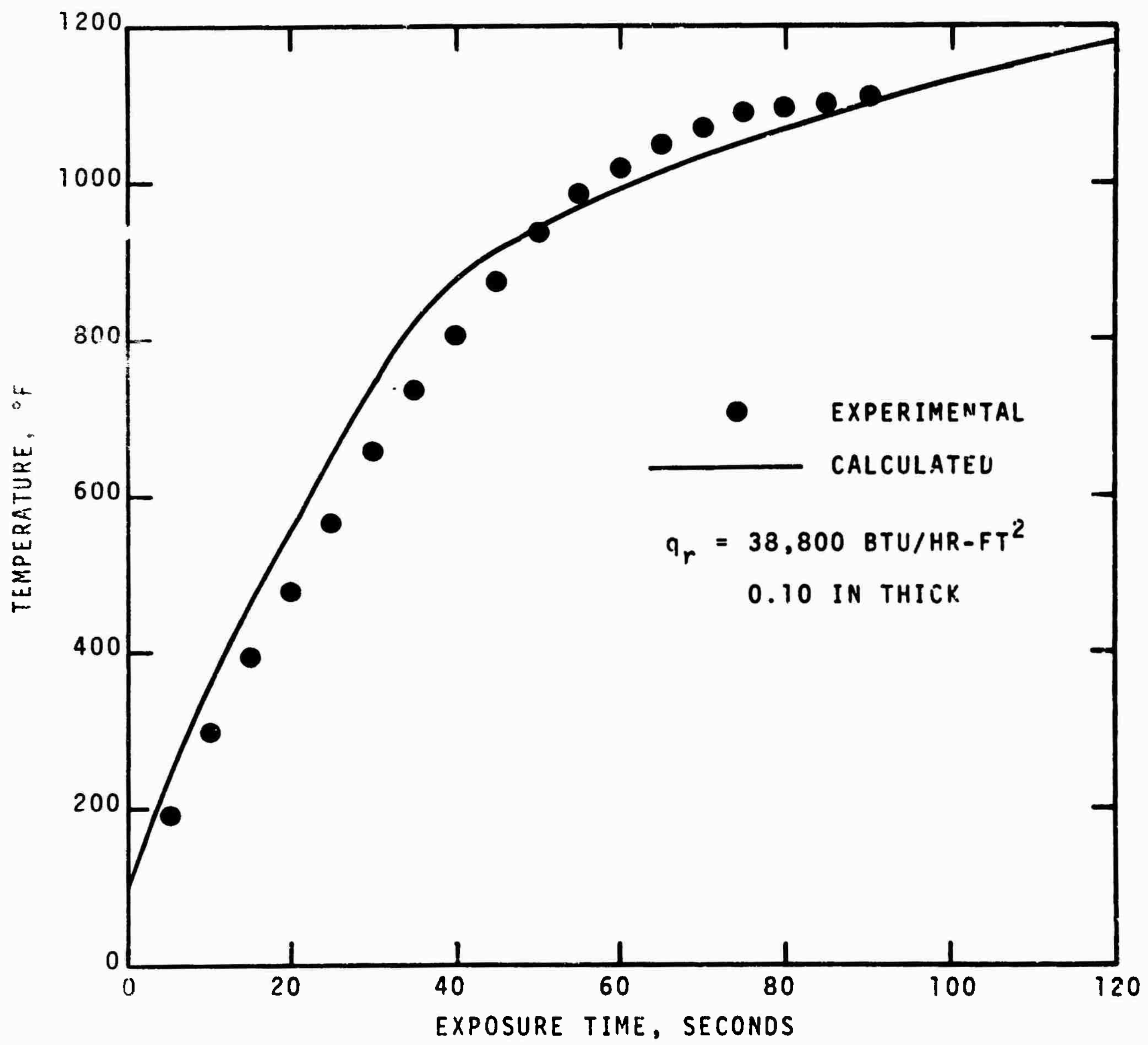

Figure 10. Calculated and Experimental Temperature Rise of 0.10-Inch Thick Aluminum.

The aluminum sample was backed on the unexposed side by insulation to minimize heat loss. The average absorptance of the exposed surface was taken as 0.8 based on spectral reflectance data. The emittance was assumed to be 1.0 because of the relatively low plate temperatures and the low spectral reflectances of the paint measured at longer wavelengths. The convective heat transfer coefficient was based on steady state heat transfer data. Agreement of calculated and measured heat fluxes is quite good for the data of Figure 9. The beginning of melting, which was taken to be about 900°F in the model, was not reached during the test.

The radiant flux in Figure 9 is well below the maximum found inside a fire. Figure 10 shows calculated and experimental results for an incident radiant flux of 38,800 Btu/hr-ft^2. The results show a little poorer comparison, with the calculated temperature rising a little faster than the experimental temperature initially, but leveling off faster once melting begins. Part of the difference may be due to changes in thermal properties of the aluminum as it is heated, and part may be due to the assumption in the calculations of linear heat absorption during melting. The important result is that the melting (or failure) time of the aluminum plate occurs less than a minute after exposure if the 0.10 in thick plate is unprotected or uncooled. This calculation illustrates the validity of the calculational procedures presented in the mathematical models in the earlier sections of this report.

During the direct flame contact heat transfer experiments the temperature of a small cylindrical thermocouple well adjacent

to the heat transfer probe was monitored continuously. (The steady state temperatures of the small cylinder were used to calculate the "flame temperature.") The temperature rise of the cylinder was calculated using an equation similar to Equation 35, but it was modified by deleting the melting term and correcting for the different area to volume ratio of the cylinder. Figures 11 through 13 show the results of the calculations and compare them with the measured temperature rise data. Figure 11 is for the nominal 12-inch fires, and shows the poorest agreement of calculated and measured results. Figure 12, for 18-inch fires and Figure 13, for 24-inch fires, show better comparison. The calculated temperature always rises faster than the measured temperature. The primary reason for the faster rise is that the calculated temperature rise is based on the maximum fire size being reached instantaneously. In the measured temperatures the fire was ignited and a few minutes heating time was required for the fuel to be heated and the fire to reach maximum size. Figure 13 also shows one heating curve where the fire became unstable and leaned over during the warmup period. The fire was stabilized and the thermocouple well was then heated to its steady state temperature. The calculations assume that the cylinder has a uniform temperature throughout. However, since stainless steel has a relatively low thermal conductivity (compared to aluminum, for example) the interior temperature may be 20 to 40°F lower than the surface temperature. This difference is a minor factor in many heat transfer considerations because it seldom limits the heat transfer rate to a significant degree.

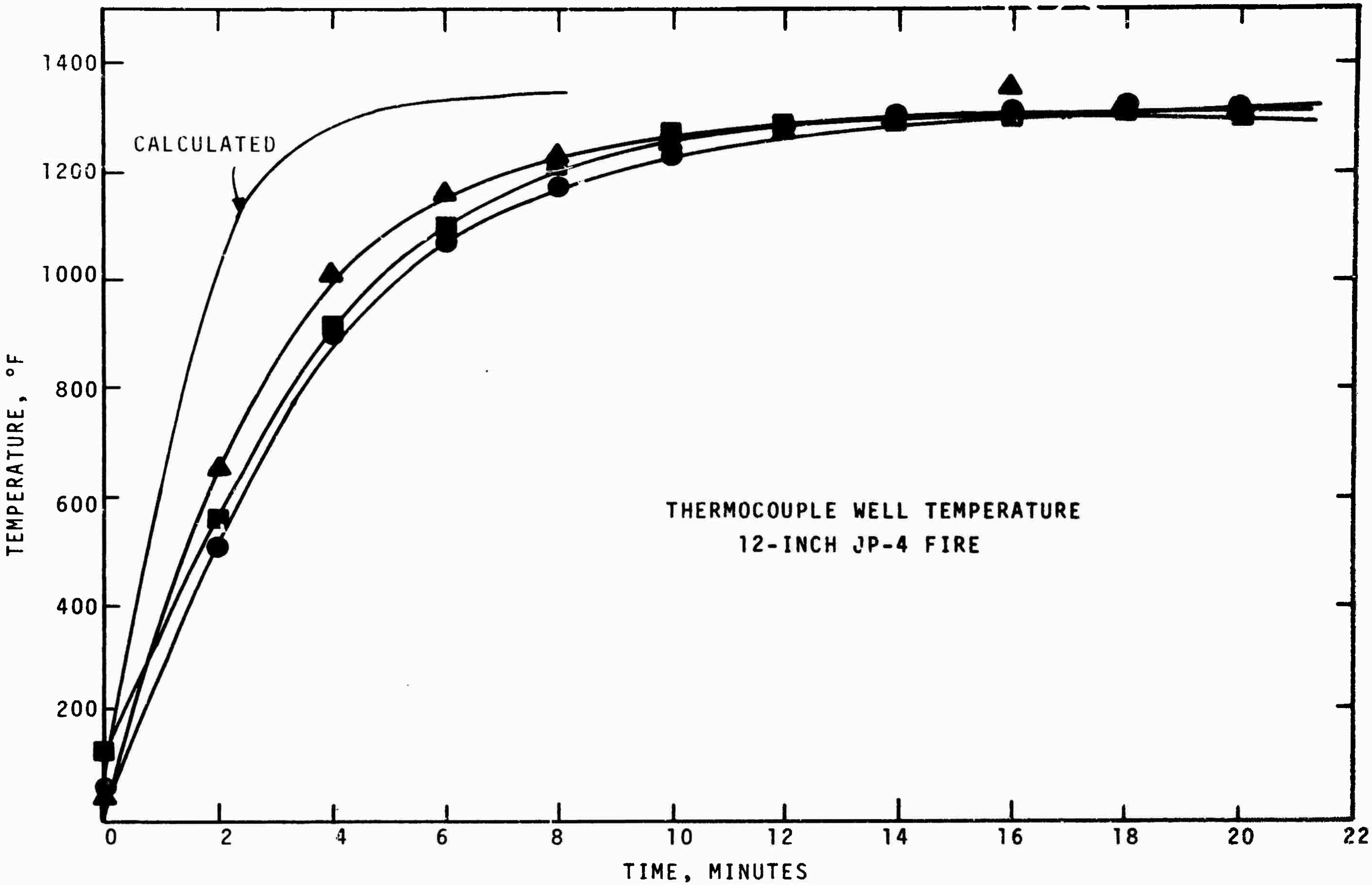

Figure 11. Thermocouple Well Temperatures for 12-Inch JP-4 Fires.

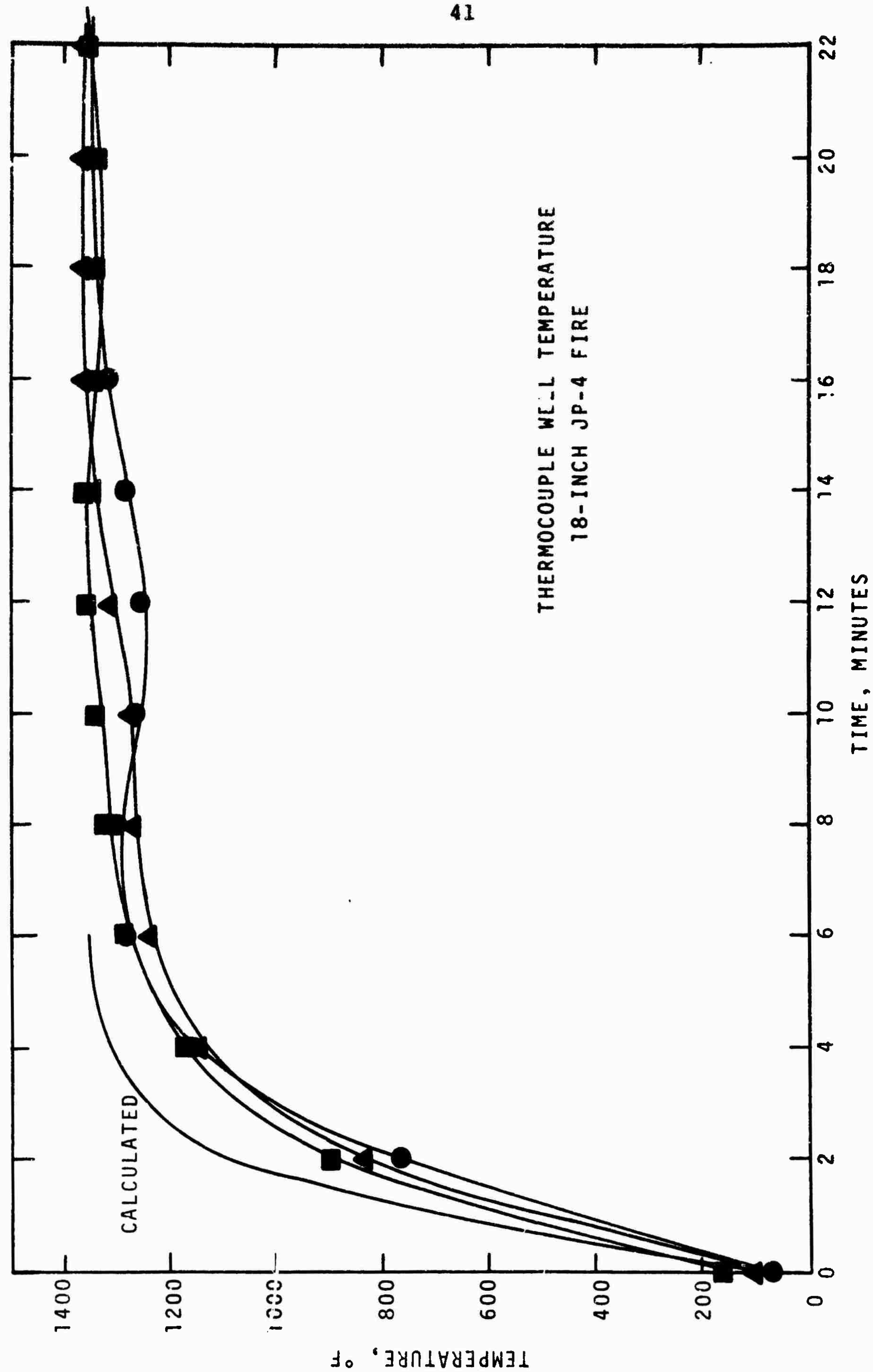

Figure 12. Thermocouple Well Temperatures for 18-Inch JP-4 Fire.

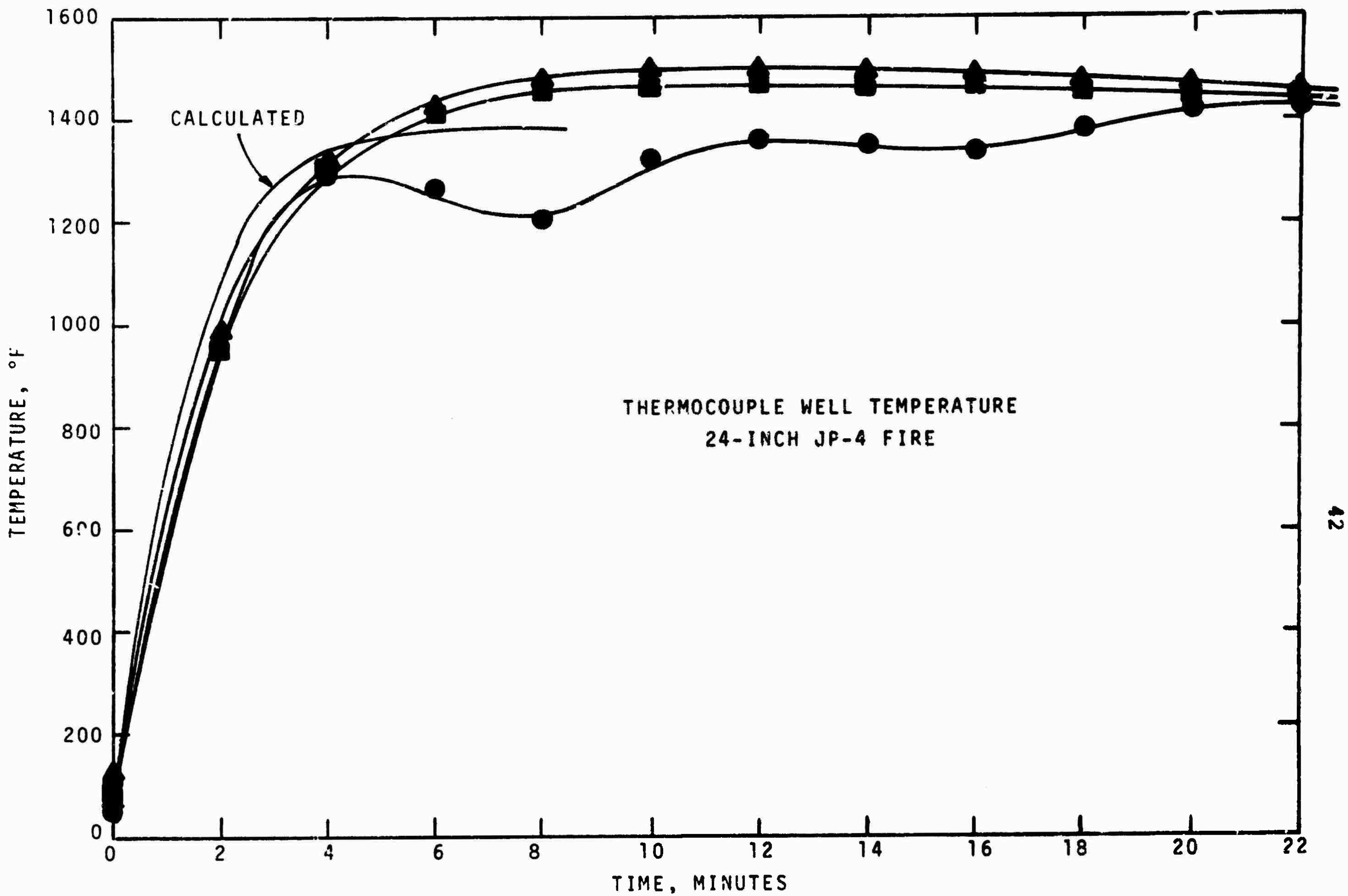

Figure 13. Thermocouple Well Temperature for 24-Inch JP-4 Fire.

Several aluminum plates were coated with intumescent paint and exposed to constant radiant heat fluxes. The aluminum temperature was measured continuously. Figure 14 shows the results for white paint on a 0.25-inch thick sample. The solid line was calculated as described previously for other aluminum samples. It agrees fairly well with the measured temperatures for an aluminum sample painted with ordinary PPG interior white latex paint. Samples painted with both PPG and Sherwin-Williams intumescent paint had slower temperature rise rates. The foaming action of the PPG intumescent paint occurred at about 280°F, and the Sherwin-Williams sample foamed at about 330°F. The PPG sample heated more slowly at first, but the foam apparently was a little less effective than the Sherwin-Williams foam at higher temperatures. The PPG sample temperature rose faster at higher temperatures.

The temperatures at which foaming occurred for the intumescent painted aluminum samples was not consistent, even for the same paint. Foaming occurred as low as 280°F and as high as 540°F, depending on the brand of paint and the pigments used for coloring. Both of the paints tried were designed for interior building use, and there are probably other intumescent paints which will give better protection for use on metals. If protection is desired for tanks, the foaming and insulating reaction must occur at sufficiently low temperatures to prevent boiling of the tank contents. The foam layer must then withstand the high temperature of the fire if it is to insulate the tank after foaming has occurred.

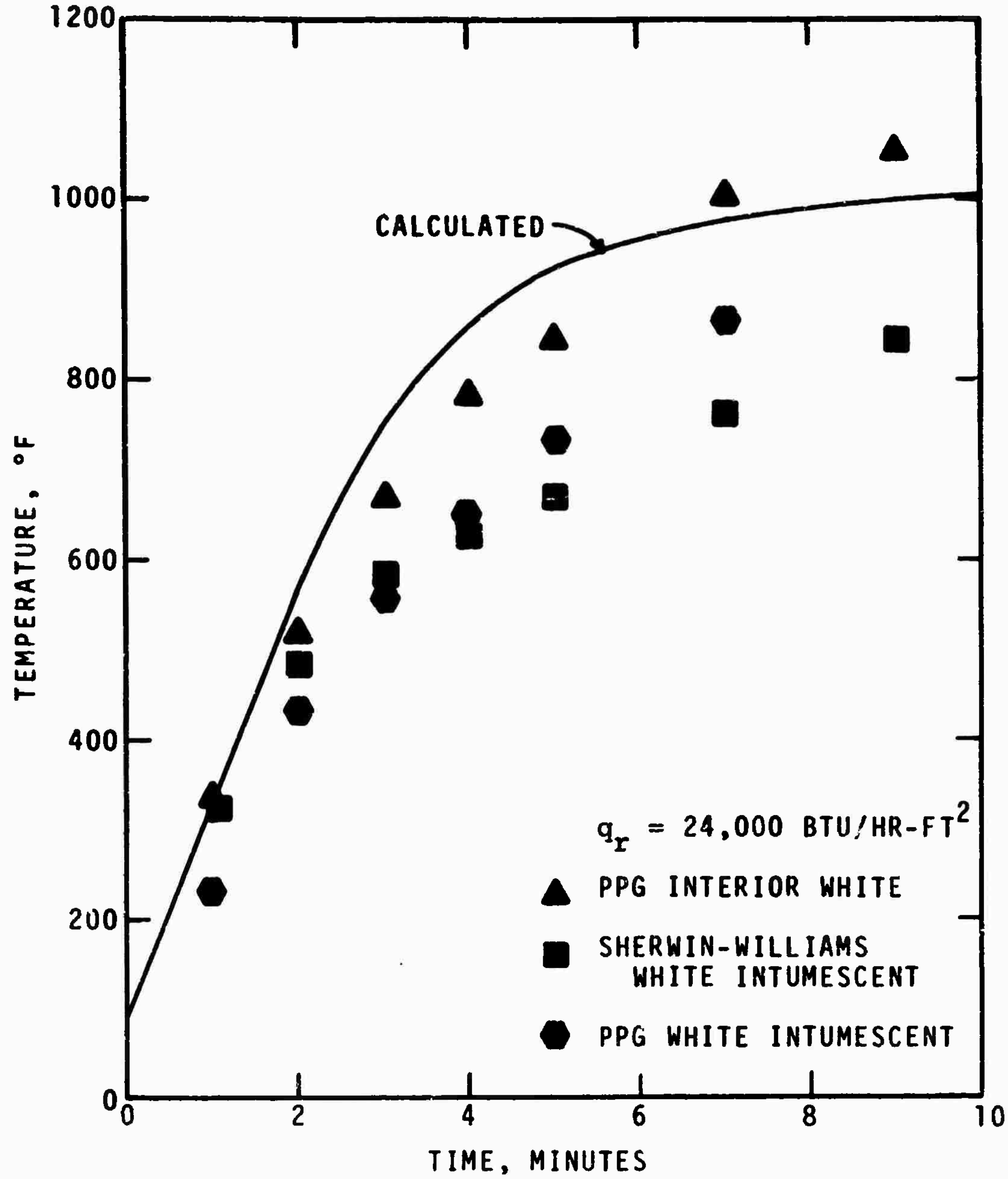

Figure 14. Comparison of Temperature Rise of Plain and Painted Aluminum Plates.

A few tests were run on the ignition time of wood samples protected with various paint combinations. The base wood samples were pine, one-half inch thick, and the samples were dried before the piloted ignition tests. Table 3 summarizes the results.

TABLE 3

SUMMARY OF PILOTED IGNITION TIMES OF DRIED PINE SAMPLES AT 20,000 BTU/HR-FT^2

Sample Preparation	Piloted Ignition Time, sec
Unpainted	14
Flat White PPG Latex	47
PPG Intumescent Paint with Carbon	113
PPG White Intumescent Paint	288
Sherwin-Williams Intumescent Paint with Carbon	47
Sherwin-Williams Intumescent Green	71
Sherwin-Williams Intumescent White	75

The results in Table 3 have been corrected to an incident radiant flux of 20,000 Btu/hr-ft^2 according to the correlation of Wesson, et al. (12). The degree of protection offered the wood sample by the intumescent paint can be significant. For example, the PPG white paint protected a wood sample well enough that ignition required twenty times as long as for an unpainted sample. The addition of carbon (from benzene soot) decreased the protection time by at least a factor of two for either paint. In general the paints foamed more vigorously and produced thicker insulating layers on wood than they did on metal. The protection they offered wood was therefore somewhat better than that for metal.

TEST PLAN

The work proposed by these fire tests is designed to provide confirmatory measurements of the heat transfer rates from large fires to a tank completely surrounded by the fire. Three fire tests are proposed, each using a tank nearly filled with water. The temperature rise of the water will be used to measure the heat transfer rate to the tank contents. The sizes of the tanks to be used will depend on their availability. The largest tank should contain about 10,000 gal, and the two smaller tanks should ideally contain about 2500 gal and 500 gal in order to obtain a good distribution of tank sizes.

Each of the three fire tests will consist of one large tank plus several smaller test panels or containers. The large tank will be situated in a test pit approximately 5000 ft^2 in area. The tank must be located in the test pit so that the flame zone is at least 15 ft thick between the tank wall and the outside of the fire or other test equipment. Ideally, the tank should also be at least 15 ft above the liquid fuel surface during the tests. Practically, it may not be possible to mount the tank 15 ft above the fuel surface and still keep it within the flame zone, unless a larger fire is used. In addition, the ambient wind must be less than about 10 mph or the flame will lean to such an extent that full coverage of the test vessel will not be obtained.

The tanks will be instrumented with thermocouples to monitor both the tank wall temperature and the temperature of the water inside the tank. Up to 12 thermocouples will be used for these measurements. In addition, two heat flux transducers will be mounted on or near the tank. One of these will measure only the radiant heat flux and the other will measure both radiant and convective flux. The measurements from these transducers can be used for comparison with the test results from the tank. The fires will burn about 20 min and provide a temperature rise of about 100°F in the tank. Tank pressure will be monitored continuously with a pressure transducer.

The tanks to be used during the test program will be chosen by the Committee and furnished either by the Committee or by other interested parties.

Several smaller test panels or containers will be used during each test. These must be positioned so that they do not interfere with the tank fire tests, but at the same time, they must also receive full heat from the fire. Each test panel will be designed to provide support for one or more experiments in which a topic of interest is investigated. For example, one panel might contain a number of samples coated with intumescent paint to measure the degree of protection that they might offer during a fire. Another panel might contain insulated samples to compare various forms of insulation. Yet another test panel might support a small-scale heat transfer probe, which could be used to aid in determining whether the size of the fire is

important in scaling convective heat transfer rates. Other experiments suggested by the committee could also be tested on the small test panels.

Four radiation heat flux transducers will be placed outside the fire. One of these will be restricted to a narrow view angle to measure the maximum surface flux from the fire. The other three will be wide angle transducers and will measure the incident flux at some location outside the fire. These data will be used to support prediction of heat transfer rates to objects outside the fire.

If the Committee decides that a full scale test using a vented tank which contains a flammable liquid is necessary, the test will be run following completion of the three tests using the water-filled tanks. The test setup would remain essentially the same, except that the duration of the test would have to be about three times as long to ensure operation of the venting device. For such a test, fuel would either have to be supplied to the test area continuously or the test pit would have to be about three feet deep. The logistics of running such a test become more difficult as the duration of the test increases, primarily because of the fuel supply problem.

The size of the fires, their long duration with respect to ordinary test fires, and the costs of building fire test grounds make it imperative to attempt to conduct the full scale tests at an existing fire testing facility. There are several test facilities that might be used, with the approval of the facility owners, some

private and some government-owned. It is anticipated that the companies or agencies owning the fire test facilities would cooperate in the fire tests since they have been willing to do so in the past. Actual negotiations with them must wait until formal approval of the program. The cost estimate for the field tests is submitted separately.

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